# SRTM Mast Damping Subsystem Design and Failure Investigation

Jeffrey W. Umland*

**Abstract**

A mast vibration damping system was developed for the Shuttle Radar Topography Mission (SRTM). The damping system development is considered from both a system perspective, and a detailed mechanism design viewpoint. The requirements derivation approach is presented, starting from the general instrument requirements, and proceeds to the determination of specific mechanism design requirements. Key component failure modes and effects, as well as the design mitigations implemented, are discussed. The diagnosis of the damping system on-orbit failure is given. The root cause of the damping system failure is provided. Conclusions are drawn, to provide guidance for future damping system implementations.

## 1. Introduction

The Space Radar Topography Mission (SRTM), illustrated in Figure 1, flew in February 2000 on the space shuttle Endeavor as the primary payload for STS-99. The objective of this joint project between the National Imagery and Mapping Agency (NIMA) and the National Aeronautics and Space Administration (NASA) was to generate a near-global high-resolution database of the earth's topography. This mission made use of Interferometric Synthetic Aperture Radar (ISAR) to digitally survey the Earth's surface from space. The primary product of this 11-day mission is a topographic database of 80% of the Earth's land surface, i. e. most land surfaces between $\pm$ 60° latitude. The resulting digital terrain data set provides a significant improvement over currently existing global topography data sets.

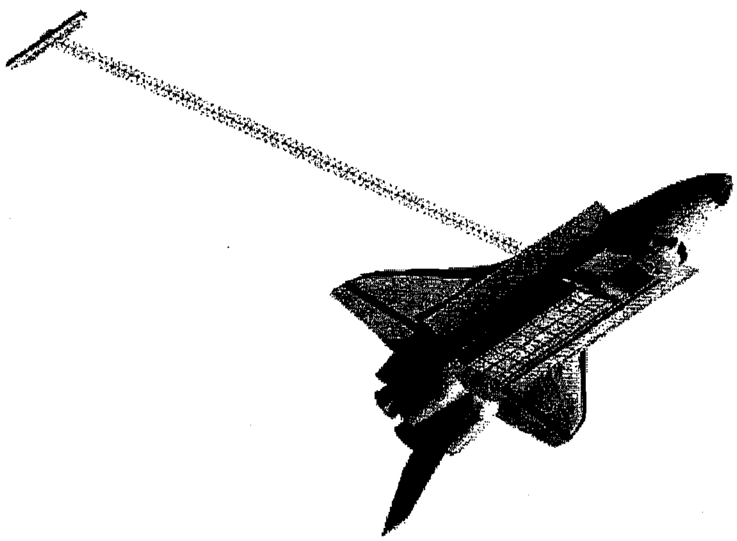


**Figure 1. SRTM Mission Configuration**

### 1.1 Instrument Overview

The SRTM architecture is based upon the Spaceborne Imaging Radar/X-band Synthetic Aperture Radar (SIR-C/X-SAR) instruments that flew twice on the Space Shuttle Endeavor in 1994, see Jordan et al, 1995. The SIR-C/X-SAR project was a collaborative effort between NASA, which developed SIR-C, and the German and Italian Space Agencies, which developed X-SAR. The SIR-C instrument was two separate SAR's, which operate in the C, and L-bands. The X-SAR instrument operates in the X-band. The combined SIR-C/X-SAR instruments including electronics essentially fill the shuttle payload bay. The primary objective of the SIR-C/X-SAR missions was the radar imaging of select "supersite" targets. SIR-

*Jet Propulsion Laboratory, California Institute of Technology
4800 Oak Grove Drive, Mail Stop 301-320
Pasadena, CA 91109

*Proceedings of the 35th Aerospace Mechanisms Symposium, Ames Research Center, May 9-11, 2001*

C/X-SAR's secondary objectives, which enabled SRTM, included the demonstration of repeat pass interferometry and scan-SAR. The repeat pass interferometry data is then used to recover the topographical features of the target surveyed. Scan-SAR is a method of beam steering that is then employed by SRTM, in the C-band, such that the radar swath width is sufficient to achieve complete mapping coverage in 159 orbits. See Rosen et al for a detailed treatment of Synthetic Aperture Radar Interferometry.

The modifications to the existing radar instrument required to collect the interferometric data included the addition of a second C-band antenna, a 60-meter mast, metrology, and additional avionics. Further, the German Space Agency provided a second X-band antenna. The fundamental SRTM instrument configuration is illustrated in Figure 1. Simplistically, SRTM makes use of two radar antennas separated by a fixed distance, or baseline, to form a fixed baseline interferometer. The in-board antenna, relative to the Orbiter payload bay, is used as both a transmitter and receiver, while the outboard antenna is only a receiver.

One of SRTM's significant features is the use of a 60-meter long deployable mast that serves to deploy an outboard antenna and create a stable baseline. The 60-meter deployable truss and its deployment mechanisms are described by Gross and Messner, 1999. An illustration of the various components which comprise SRTM is given in Figure 2. The structural dynamic issues associated with a 60-meter mast and large tip mass, i. e. the outboard antenna, deployed from the Shuttle required significant attention during the design and implementation of SRTM. Further, SRTM implemented a mast vibration damping system specifically to meet certain mast dynamic motion constraints, as well as to supplement the Orbiter reaction control system with regard to control system stability. The topic of this paper is the design of the mast vibration damping subsystem. Further, the mast vibration damping subsystem failed to function on-orbit, hence the failure diagnoses that occurred both during the mission, and post-mission are discussed.

## 2. Mast Vibration Damping System

The mast vibration damping requirements were driven by several coupled factors, specifically: a) the metrology system's mast dynamic motion tracking capability, b) SRTM instrument pointing requirements,

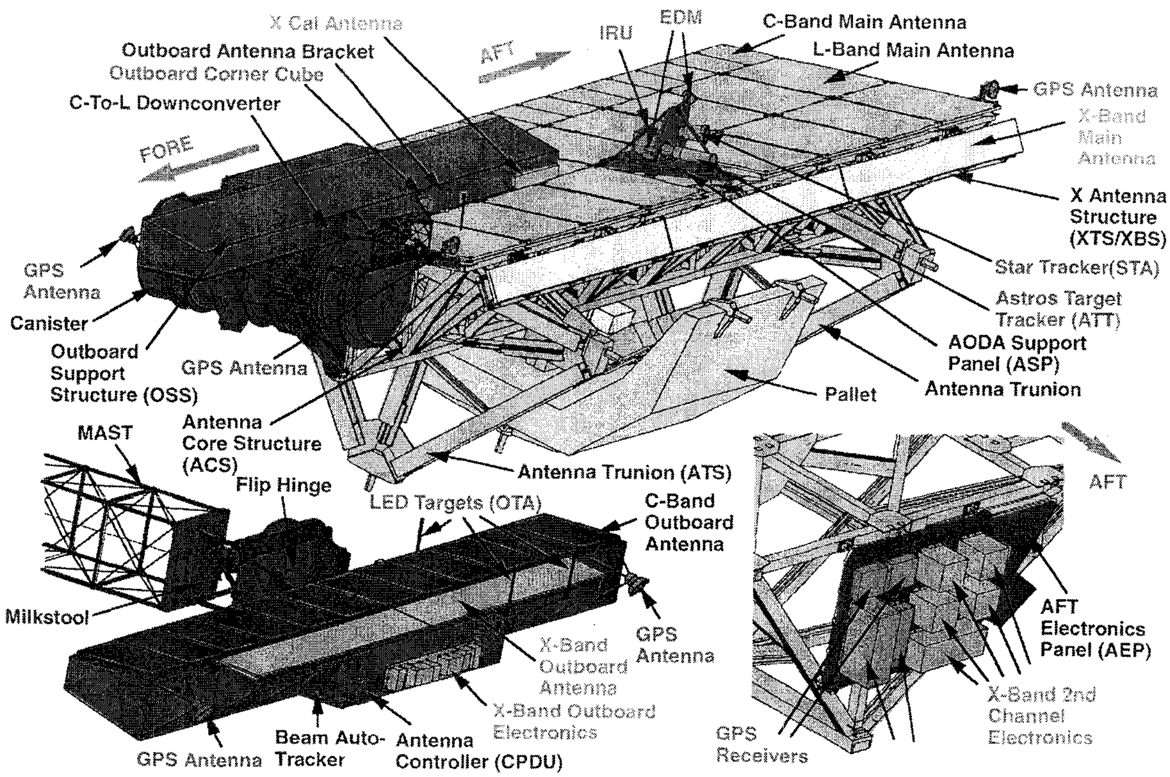



**Figure 2. SRTM Instrument Component Layout**

c) attitude control disturbance torques associated, and d) the Orbiter dynamic disturbance environment induced by the attitude control system. SRTM utilized a metrology system to provide relative outboard antenna position and attitude knowledge during radar interferometer operation. This metrology system consisted of two subsystems. The subsystem, which drove the damper design, consisted of the Astros Target Tracker (ATT), and three Optical Target Assemblies (OTA's). The ATT was essentially a star tracker that had been refocused to 60-m. Each OTA contained an LED, which was pointed towards the ATT, and acted as a psuedo-star that the ATT was able to track. The combination of the ATT and OTA's provided an accurate estimate for five of the outboard antenna's six rigid body degrees of freedom, the ATT and OTA constellation does not accurately measure range to the outboard antenna. A Leica range finder supplemented the ATT by directly measuring the distance to a retroreflector array mounted on the outboard antenna. With respect to instrument pointing, the nominal attitude during data acquisition was to: 1) fly the Orbiter with its tail pointed along the velocity vector, 2) rotate the Orbiter about its roll-axis such that the mast was 45 degrees from the local vertical, and 3) radar radiating surfaces oriented toward the ground. The Orbiter reaction control system was used to maintain SRTM pointing within a 0.01 deg attitude deadband, and 0.1 deg/sec attitude rate deadband. Given the attitude requirement and combined SRTM/Orbiter mass properties, the gravity gradient torque was the dominant disturbance torque to the Shuttle reaction control system. Specifically, the gravity gradient torque tended to rotate the combined Orbiter and mast system such that the mast longitudinal axis was oriented along the local vertical. The Orbiter's Digital Auto-Pilot (DAP) was configured such that the 24 lbf. Vernier Reaction Control System (VRCS) jets were used for attitude control during radar operations. Based on the attitude control requirements, and configuration versus the disturbance torque applied to the system, the reaction control system generated positive roll commands that resulted in jet firings to counter the gravity gradient torque. As a consequence of these jet firings transient vibrations in the mast were generated. Mast tip motion was not a concern to the operation of the radar as an interferometer provided that the knowledge of the tip motion was acquired. Hence, the ATT and LED's were added to track the mast motion. The capability of the ATT to acquire and track the motion of the LED's defined a maximum rate of mast motion that could be tracked, this limit was defined to be 2.36 in/sec (6 cm/sec) at the tip of the mast. Additionally, in order for the ATT to acquire the LED the mast tip rates were required to be less than 0.24 in/sec (6 mm/sec) for ten percent of the time during data acquisition. Therefore a mast vibration damping system was implemented to enable the ATT to acquire and track the mast motion given the vibration environment generated by the attitude control system.

### 2.1 Vibration Damping Requirements

SRTM implemented a mast vibration damping subsystem in order to meet mast tip dynamic motion requirements. Early in the project design phase, a preliminary coupled Orbiter and deployed mast structural dynamic math model was created. This math model was employed in attitude control simulations to estimate: a) propellant consumption estimates, b) mast tip dynamic motion estimates, and c) mast damping requirements. Given that the final damping system implementation was not determined, the preliminary math model was a "modal" model, i. e. the true complex modal behavior associated with discrete viscous damping elements was approximated. The results of these early simulations showed that the mast damping mechanisms should be designed to achieve "high", i. e. greater than 10%, damping ratios in the deployed mast's first orthogonal bending vibration mode pair and the first torsional vibration mode.

### 2.2 Vibration Damping System Concept

Conceptually, the approach employed towards the design of the mast damping system was to concentrate sufficient modal strain energy at the mast interface to the inboard antenna such that only a few discrete damping elements are required to damp the mast. In practise, what this means is that the structural elements which connect the mast to the to inboard antenna structure were softened, i. e. their stiffness was reduced, such that approximately half the modal strain of the deployed system's first modes of vibration was concentrated at these elements.

The structural attachment of the mast, via the mast canister, to the inboard antenna structure is shown in Figure 3, this structure was called the canister attachment truss (CAT). The CAT is a kinematic, i. e. statically determinate, structure which serves to attach the mast, and the mast canister to the inboard antenna structure. The CAT forward bipod and the CAT axial strut form a rigid tripod with a monoball, or spherical bearing, at its vertex. The monoball located at the vertex of this rigid tripod is a fixed rotation point about which the entire deployed mast rotates as a rigid body. The attitude of the mast relative to its fixed rotation point is controlled by the aft bipod, and the vertical strut. The mast damping elements were

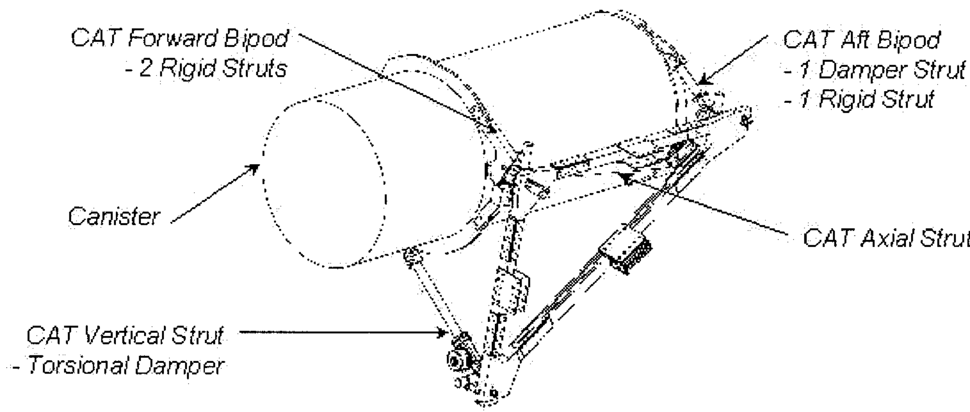



**Figure 3. Mast Canister and Canister Attachment Truss (CAT)**

located at the aft bipod, and the vertical strut. The damping elements located in the aft bipod controlled the mast's first two orthogonal bending modes, while the damping element located in the vertical strut controlled the mast's first torsional mode. A conceptual model of the damping system implementation is given in Figure 4.

Note that the CAT is the only structure, which connects the SRTM outboard equipment, i. e. outboard antenna, mast, and mast canister, to the Orbiter. The mass of the outboard equipment is on the order of 3600 lbm. Hence, the CAT was designed to meet structural requirements derived from launch and landing, i. e. specific loads and frequency requirements. Conceptually it was acknowledged that the damping mechanisms embedded within the CAT would be required to be locked, i. e. "stiff", for launch and landing, and unlocked, i. e. "soft", during on-orbit operations.

2.3 Vibration Damping Mechanism Design

Two distinct damping mechanisms were developed for SRTM. The bending mode damping mechanism is shown in Figure 5, this mechanism is also the CAT aft bipod. The aft bipod upper strut is rigid while the lower strut contains three relatively soft springs and three viscous damping elements. The viscous damping elements are mechanically in parallel with the springs. Additionally, a caging mechanism is employed to lock out the soft springs for launch and landing. The torsional mode damping mechanism is also the CAT vertical strut. The torsional mode damping mechanism is similar to the bending mode damper, but uses only one damping element rather than three. The torsional damping mechanism is shown in Figure 6.

The dynamic impedances of the respective damping mechanisms were sized in a two-step process. Also, given the CAT geometry, and the mast mode shapes, the sizing effort for a given damper was uncoupled from the other. First, the static spring stiffness was determined as outlined above, that is the mechanical spring stiffness of the damping mechanism was adjusted, in this case reduced, until approximately half the modal strain energy for the mode or modes of interest was concentrated in the elements which

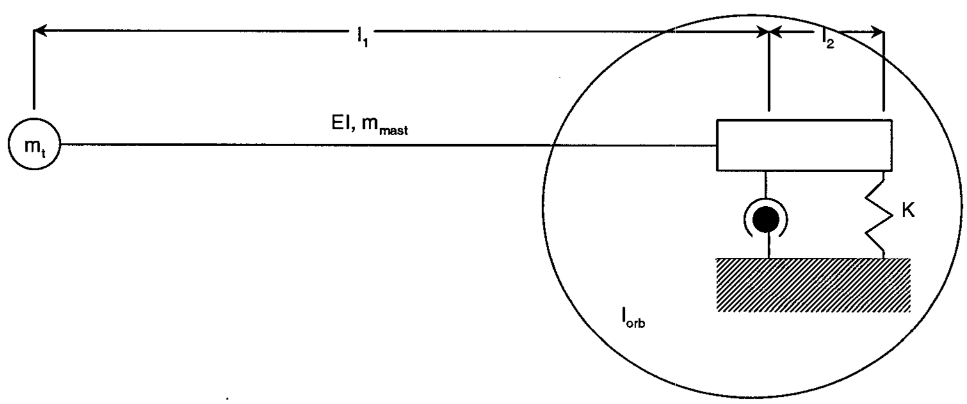



**Figure 4. SRTM Conceptual Dynamic Model**

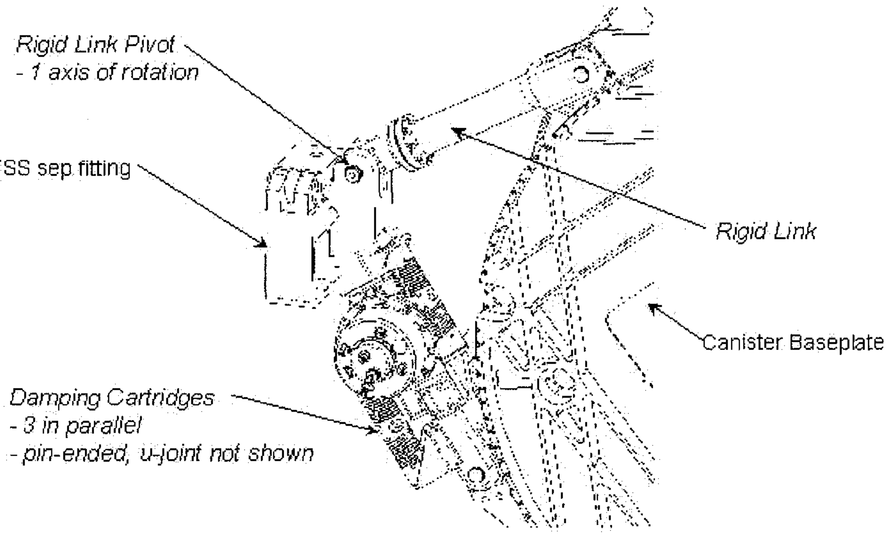



**Figure 5. Bending Mode Damping Mechanism (CAT Aft Bipod)**

represented the damping mechanism. Relative to the bending mode damper, variations on this approach were also tried and it was determined that acceptable results were achieved with the upper bipod strut rigid and the lower bipod strut soft. Specifically, the cumulative static spring stiffness of the bending mode damper lower strut was specified to be 500 lbf/in. The cumulative static spring stiffness of the torsion mode damper was determined to be 25 lbf/in. The second step in the damper requirement definition process was to determine the damping coefficient. A sensitivity analysis was performed by allowing the cumulative damping coefficient for a given damping mechanisms to be analytically varied from zero to infinity. Once the root locus associated with the system's structural dynamics was known, the mechanism damping requirement was determined by selecting the desired set of structural dynamics, i. e. frequency and damping ratio, from the root locus plot. The root locus plot for the bending mode damping mechanism is given in Figure 7. Note that in the root locus diagram the radial grid represent contours of constant damping ratio, while circles represents frequency given in radian/sec. Further the arrow shown next to the loci implies the direction of increasing damping coefficient. The cumulative damping coefficient for the bending mode damping mechanism was determined to be 2100 lbf-sec/in, implying that the damping coefficient for each of the three damping cartridges is 700 lbf-sec/in. While the damping coefficient for the torsion mode damping mechanism was specified to be 25 lbf-sec/in.

2.3.1 Damping Cartridge Design

A trade study was conducted early in the SRTM design phase, first to determine the necessity for a mast

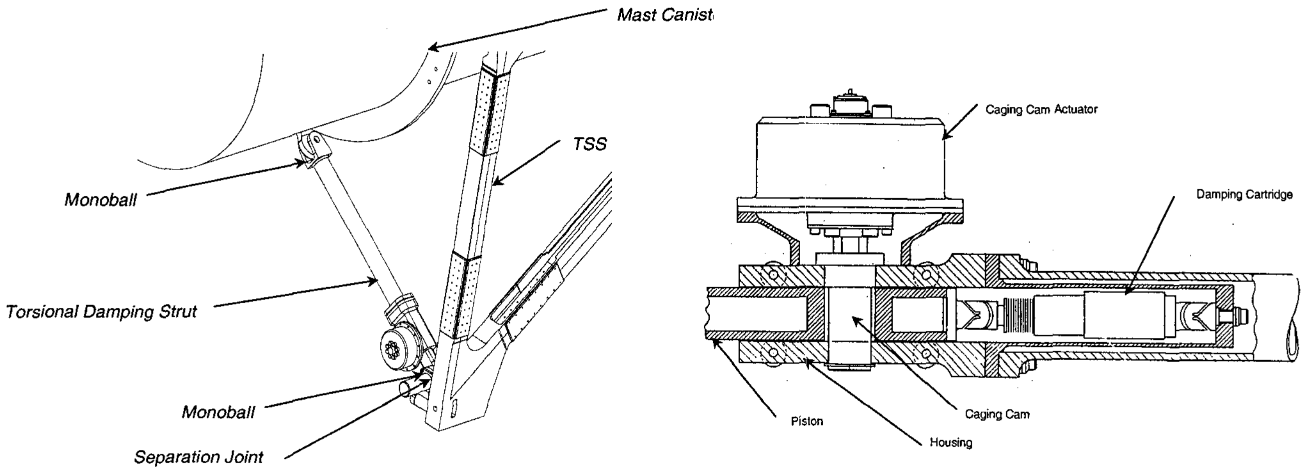



**Figure 6. Torsion Mode Damping Mechanism**

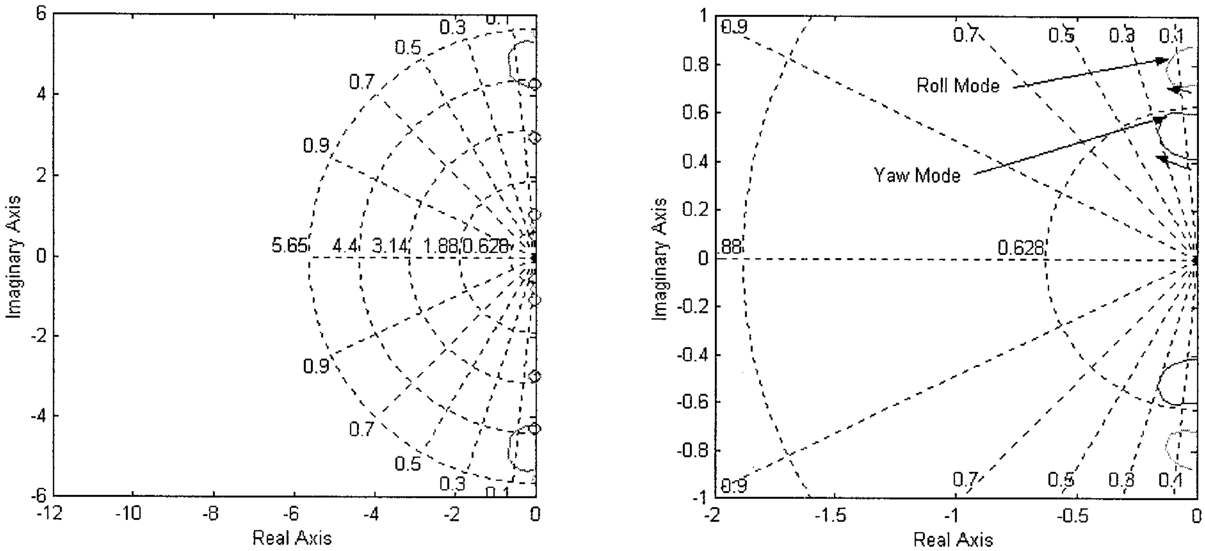



**Figure 7. Bending Mode Damping Mechanism Root Locus Diagram**

damping system, and then to determine an adequate damping system design concept. The mast damping approaches considered in the trade study included: active piezoelectric struts, Coulomb friction dampers, and linear viscous dampers. A linear viscous damping approach was selected based on several factors: 1) passive energy dissipation, 2) simple design concept, 3) amenability to existing analysis tools, and 4) design heritage. The viscous damping cartridges selected and used in both SRTM mast vibration damping mechanisms were a modification of a hermetically sealed vibration isolator previously described by Klembczyk and Mosher and U. S. Patent 4,638,895. The viscous damping cartridge contained within the vendor's vibration isolator readily met the SRTM damping approach selection criteria. It was expected that several modifications of the existing damping cartridge design would be needed in order to meet SRTM requirements, specifically the physical damping coefficient, and the overall stroke, i. e. length of travel. There are several differences between the SRTM damping cartridges as procured and the vibration isolator described by Klembczyk and Mosher. First, both the helical spring and the universal, zero friction flexure, were not included as part of the damping cartridge definition, and hence part of the damping cartridge contract. Further, as the damping cartridge orifice matured it was determined that the orifices through the damping head were not required; rather the annular orifice between the damping head and the cylinder was sufficient.

The principles, which govern the damping cartridge operation, are straightforward. A damping head is placed within an essentially closed fluid filled cylinder. When the damping head is translated relative to the cylinder, a viscous shear stress is generated across the fluid that exists within the annular orifice created by the clearance between the damping head outside diameter and the cylinder head inside diameter. Wide ranges of viscous damping coefficients are achievable given such design variables as fluid viscosity, and annular orifice size. The damping head is supported on a piston rod. A set of labyrinth seals, which support the piston rod relative to the cylinder, provide two critical functions for proper damping cartridge operation. First the labyrinth seal is a dynamic seal, i. e. an infinite fluid resistance, such that as the damping head is translated in the cylinder, fluid flows between the two fluid chambers created on both sides of the damping head, and not through the annular orifice created between the piston rod outside diameter and the labyrinth seal inside diameter. Secondly, the labyrinth seals act as linear journal bearings such that linear motion of the piston rod relative to the cylinder is possible. Finally, the entire unit is hermetically sealed. A pair of welded metal bellows is used as flexural seals such the entire damping cartridge is hermetically sealed relative to the external environment. The two fluid chambers created by the bellows assemblies are connected via a crossover port, such that the required fluid flow between these two chambers is accomplished. The fluid resistance of the crossover port is negligible.

The driving requirements for the SRTM damping cartridges are given in Table 1. The final damping cartridge designs for the bending and torsion mode damping mechanisms were mechanically identical. The only real difference between the two damping cartridges is that the torsion mode damping cartridge was filled with 10 cSt silicone fluid, while the bending mode damping cartridges were filled with 100 cSt silicone fluid.

During the damping mechanism design phase, possible damping cartridge failure modes were considered, as well as their effects on the overall system structural dynamics, the failure mode credibility assessed, and required failure mode mitigations were identified. Specifically, three cartridge operational states were determined: 1) nominal operation, 2) failure to a "soft" condition, and 3) failure to a "stiff", or seized, condition. Nominal cartridge operation was defined in the sense that, as the cartridge was "stroked", i. e. the damping head translated relative to the cylinder, the force required for relative motion was proportional to the rate of relative motion. Dispersions allowed for under nominal operation included a ten percent absolute tolerance on the cartridge physical damping coefficient such that the variation of the damping coefficient given the operating temperature range requirements was accounted for in the structural dynamic and attitude control assessments.

The "soft" failure condition was a generic failure mode created to describe the situation when the damping cartridge stroked readily, but failed to generate a damping force, i. e. a force that is proportional to the rate of relative motion. A hypothetical example of this failure mode could be assumed should the damping cartridge fluid leak from the cartridge through a weldment crack. A damping cartridge "soft" failure due to a weldment failure was considered credible based on several factors inherent to the cartridge design and worksmanship verification. X-ray inspection of all cartridge weldments was considered inconclusive relative to weldment workmanship, given that any conclusion regarding the integrity of the convoluted bellows inside diameter welds would be speculative at best. Standard leak tests were performed on the bellows assemblies to verify hermeticity, but a nondestructive evaluation of the bellows inside diameter welds, for a measurement of such parameters as weld penetration, was considered impractical. Further, standard leak test evaluation of the cartridge final closure welds was also impractical. Though the final closure welds were visually and X-ray inspected. The soft failure of a damping cartridge was considered credible. Additionally, this failure mode was considered to be a random failure, rather than a failure common to all damping cartridges implemented on SRTM. Based on the soft failure mode coupled with a Shuttle payload safety two-fault tolerance requirements, in this case the concern was changes in the structural dynamic which could lead to attitude control instability, the bending mode damping mechanism was designed to incorporate three damping cartridges.

Gas bubble formation within a fluid filled damper is a significant concern; this concern is further exacerbated in space applications. Stewart, Powers, and Lyons, 1998, have discussed an example of this problem, in regard to rotary dampers. This degraded performance condition is similar to the soft failure mode. Effectively, a deadzone or backlash is created within the damper when a gas bubble is formed within the fluid. The primary protection against gas bubble formation within the viscous fluid used here was a volumetric overfill, and the use of the bellows as an accumulator, such that pressure was maintained on the fluid once the cartridge entered a vacuum environment. The approach taken toward the over pressurizing the fluid was to fill the damper at a temperature below the specified operating temperature, in this case below -40ºC. The silicone fluid has a positive volumetric expansion coefficient relative to temperature, i. e. as the fluid's temperature increases its volume increases as well. Further, in the damping cartridge application considered here the bellows were used as an accumulator, that is the bellows were a flexible container which could expand and contract with the fluid's volumetric changes. Given that the bellows are elastic with volumetric spring stiffness, then as the fluid and bellows expand

**Table 1. Damping Cartridge Requirements**

| Parameter | Bending Mode Damping Cartridge | Torsion Mode Damping Cartridge |
|---|---|---|
| Stiffness (lbf/in) | 25 | 25 |
| Damping Coefficient (lbf-s/in) | 700 | 25 |
| Stroke (in) | 0.860 | 0.860 |
| Frequency Range (Hz) | 0.05 – 2 Hz | 0.05 – 2 Hz |

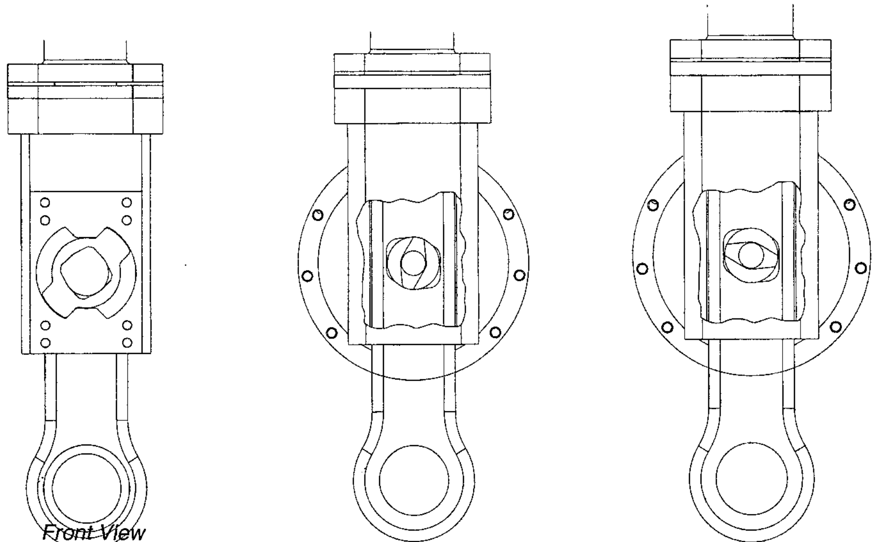
due to a temperature increase, moderate fluid pressure increases are incurred due to accumulator effect of the bellows. Finally, thermal control was applied to the damping cartridges such that the nominal minimum expected operating temperature was -17ºC (0ºF). Given this minimum expected operating temperature, the fluid volumetric expansion, and the bellows spring stiffness, the minimum expected fluid pressure was determined to be approximately 15 psi.

The silicone fluid was degassed prior to cartridge filling. Unfortunately, the fluid was only exposed to a vacuum of approximately $10^{-2}$ Torr. Further, the fluid was exposed to the ambient atmosphere during the filling process. Hence, the fluid was not considered completely degassed. Therefore, the possibility of bubble formation was considered highly unlikely, but credible. The issue of gas bubble formation within the damping cartridge was ultimately resolved by requiring a set of structural dynamic identification tests be performed during that SRTM on-orbit checkout phase. The success criteria of the identification tests was structured such that if the measured structural dynamics matched preflight predictions then the mission could proceed to its mapping phase, conversely if the measurements did not match the predictions then changes would be required to the attitude control design before mapping was authorized.

The "stiff" failure mode that generically described any of the conditions where the damping cartridge was seized, i. e. it could not be stroked. The primary concern here was seizure of the piston rod in the linear journal bearings, i. e. the labyrinth seals, due particulate contamination that could cause this single sliding surface interface to jam. The mitigations employed to protect against journal bearing seizure included silicone fluid filtration prior to cartridge filling process, cartridge component precision prior to assembly, and cartridge assembly in a controlled clean environment, i. e. a flow bench. Given that the SRTM damping cartridges were a modification of a proven design that had not exhibited any in service seizure type failures, the design heritage provided additional confidence that this design was not susceptible to seizure type failures. In an effort to minimize any friction force due to incidental contact of the piston rod with the labyrinth seals, tight concentricity tolerances were employed on the cylinder, and labyrinth seal assemblies. Further, cross-blade flexures, i. e. universal joints, were employed at each end of the cartridge such that end moments, which would be created due to various misalignments, would be minimized. Fluid freezing was also classified as a stiff failure. This type of stiff failure was protected against by thermally controlling the damping cartridges such that the nominal minimum cartridge temperature was -17ºC (0ºF) versus the silicone fluid freezing temperature of -70ºC. Further the minimum cartridge operating temperature without thermal control was -35ºC. For the purposes of the overall damping mechanism design, the damping cartridge stiff failure mode was not considered credible based on the mitigations described. On the other hand, for the purposes of the attitude control system the damping mechanism stiff failures were considered credible, this determination was not based on any concern associated with the damping cartridge.

### 2.3.2 SRTM Damping Mechanism Design

Recall that the CAT is a kinematic (statically determinant) truss. It is noted that the CAT aft bipod upper and lower struts, i. e. the bending mode damping mechanism, and the vertical strut, i. e. the torsion mode damping mechanism, are axial force elements. The structural requirements on these truss elements are dependent on mission phase. In general for launch and landing these truss elements were required to be stiff and strong. The specific launch and landing limit load requirements were 10,000 lbf, and 22,000 lbf, for the aft bipod lower strut, and the vertical strut respectively. While on-orbit these dynamic impedance requirements imply that these struts were very compliant. Hence, a common lockable linear bearing design concept was implemented for both damping mechanisms see Figure 8. Supporting a piston with a housing with cam follower bearings created a linear bearing. The piston was locked and unlocked relative to the housing via a caging cam. The caging cam is a diamond pin in effect. The caging cam was actuated by a 5000 in-lbf DC motor and gearbox assembly provided by American Technology Consortium (ATC). Knowledge of the linear bearing state, that is caged, fully un-caged, or in determinant, was provided by a set of limit switches that were actuated by mechanical features on the caging cam. The limit switches used here were the Honeywell 9HM1's. Payload safety considerations levied a two-fault tolerant requirement on knowledge of the caging cam position, and hence the linear bearing status. Therefore, two sets of three independent switches were integrated into each caging cam and actuator assembly, such that positive two-fault tolerant knowledge of the caging cam's position relative to being caged or fully un-caged was provided. Additional knowledge of the caging cam status was inferred via monitoring of the motor current draw during the act of caging or un-caging, and then comparing to similar ground test data.



Front View
-Switches cutaway
- Cam against closed stops

Back View, stop & adapter cutaway
- Cam in closed position
- Plunger locked to housing

Back View
- Cam in open position
- Plunger unlocked to housing

**Figure 8. Caging Cam Conceptual Layout**

A linear potentiometer was used to provide a relative displacement measurement for each of the two damping mechanisms. Betatronix, Inc provided the potentiometer. The damper linear displacement measurement was implemented in order to provide additional useful data regarding the state of each mechanism. Further the data obtained from the displacement sensor was considered a measure of the dynamic health of the device since this sensor was sampled at the rate of 1 Hz.

## 3. Damper Failure Investigation

3.1 Pre-Mission Investigation and Flight Rationale

Approximately two months before the eventual SRTM launch date the flight spare torsion mode damping cartridge was discovered to have seized, while the spare bending mode damping cartridge appeared to function normally. Further quantitative evaluation of both damping cartridges showed that: 1) the spare bending mode damping cartridge function normally, and 2) the torsion mode damping cartridge began to stroke only after a force of greater than 100 lbf was applied. Normally, the torsion mode damping cartridge stroked once a 0.25 lbf was applied. Additional testing showed that once the torsion cartridge was cooled to 0ºC the cartridge stroked readily. A trend was inferred based on these two data points in the sense that whatever caused the cartridge seizure was relieved as the unit's temperature was decreased. Based on this trend, a dimensional interference between the labyrinth seal inside diameter and the piston rod outside diameter was suspected to be the cause of the cartridge seizure. The piston rod was 15-5 stainless steel. While the labyrinth seal was made from Torlon material, and installed into a 302 stainless steel housing with a moderate interference fit. Based on the materials used for the labyrinth seal and the piston rod, as well as the residual stress created in the labyrinth seal due to the press fit, it was known that the clearance between the labyrinth seal and the piston rod would increase with a decreasing unit temperature. Hence, given the test data, a possible failure cause was suspected. The torsion mode damping cartridge was disassembled, and a dimensional interference of 0.0002" to 0.0003" was measured between one of the labyrinth seals and the piston rod. Further it was determined that the piston rod outside diameter had not changed and that the inside diameter of the labyrinth seal was smaller than expected. The design data on the relevant assembly drawing showed that the normal clearance between the labyrinth seal and the piston rod should be 0.0014" ±0.0002". Examination of the quality assurance paperwork for each damping cartridge set showed that a final machining operation, where the clearance between the seal and rod was adjusted within the requirements, was omitted on the entire torsion mode damping cartridge lot. The required final machining operation was performed on the bending mode damping cartridges.

Yet, the torsion mode damping cartridges were verified to function properly during acceptance testing 10 months earlier, which implies that adequate, if not the required, clearance existed between the seal and piston rod. Therefore, a very serious concern was evident in that, the labyrinth seal was dimensionally unstable with respect to time. Two possible physical mechanisms were proposed to explain the temporal dimensional instability of the labyrinth seal material: 1) swelling due to silicone fluid absorption, and 2) stress relaxation driven by the residual stress associated with the press fit. Fluid absorption into the labyrinth seal material was not considered credible given the dissimilar chemistry of the silicone fluid and Torlon. Stress relaxation of the labyrinth seal material was postulated as the physical mechanism that led to the seizure of the torsion mode damping cartridge. Included in the damping cartridge assembly procedure is a labyrinth seal post-press fit stress relief heat treatment. During this investigation it was determined that the heat treatment performed was inadequate. This determination was based on a set of tests run on residual labyrinth seal assemblies which had been previous heat treated. A second heat treatment was performed on the residual hardware and typically a 0.0005" decrease in the labyrinth seal inside diameter was measured.

The seizure of the spare torsion mode damping cartridge was attributed to two factors: 1) the clearance between the piston rod outside diameter and labyrinth seal inside diameter was less than required on the assembly drawing; and 2) the clearance between these two parts was reduced to an interference via stress relaxation of the labyrinth seal. There was no justification that the flight torsion mode damping cartridge was any different than the spare, thus it was concluded that flight torsion mode damping cartridge was likely seized as well. On-orbit operation of the torsion mode damper was not required either for instrument performance or for payload safety. On the other hand, proper bending mode damper operation was required for instrument performance. Payload safety considerations required that the bending mode damper function within an expected and previously assess envelope. An acceptable for flight rationale was generated for the bending mode damper based on: 1) quality assurance paperwork verification that the clearance between the labyrinth seal and the piston rod was per the drawing callout, and 2) an adequate clearance between the seal and piston rod was estimated based on a worse case assumption of the dimensional change of the seal inside diameter.

### 3.2 On-Orbit Failure Diagnosis

As part of the SRTM on-orbit checkout procedure, flight rules required that the natural frequencies of the deployed mast's first vibration modes be measured. The rationale behind this requirement was that the stability of the Shuttle's reaction control system is assured by proper placement of and sizing of notch filters which then serve to mask low frequency dynamics. This system identification was performed with the dampers locked, and unlocked. Additionally, the system identification was also performed for both high and low level excitation. Based on comparison of the system identification results from the dampers locked versus unlocked tests it was concluded the dampers were inoperative. Further, zero damper relative displacement was measured via the displacement sensor mounted on each damping mechanism. The dampers were re-locked, and the mission continued to a successful conclusion. Proper instrument operation was achieved without functional dampers by utilizing overlapping design performance margin contained within the other SRTM sub-systems.

### 3.3 Post-Mission Failure Investigation

Following the mission a failure investigation was conducted in order to determine the root cause of the SRTM damping mechanism failures. The conclusion of this investigation was that both damping assemblies failed due to a common mode failure attributed to the damping cartridge mechanical design. Specifically, it was found that all damping cartridges assembled for SRTM had seized. The SRTM damping cartridge seizure was traced to a dimensional interference between the piston rod outside diameter and the linear bushing inside diameter. It was further determined that the inside diameter of the linear bushing, made from Torlon, had changed dimensionally; i. e. the ID had reduced, thereby eliminating the required clearance between the bushing and piston rod. As discussed above, the two possible physical mechanisms which explain the temporal instability of the bushing inside diameter are: a) silicone fluid absorption by the linear bushing and b) long term creep of the linear bushing due to residual stress. Prior to the launch of the mission the fluid absorption of the labyrinth seal material was not considered a credible explanation for the dimensional change of the labyrinth seal material. It is noted that all the damping cartridges assembled for SRTM ultimately seized due to labyrinth seal interference with the piston rod; on the other hand, a labyrinth seal test unit that has not experienced long term fluid exposure has not seized. Therefore, it is possible that the assumption that, the dimensional instability of the labyrinth seal due to fluid absorption is incredible, is not valid. The conclusion reached to date is that the dimensional change of the labyrinth seal material is attributable to one probable cause and an additional possible cause, that is stress relaxation and fluid absorption, respectively.

A final technical conclusion reached here is that the material used for the labyrinth seal is inappropriate for this application. An understood but overlooked requirement for the labyrinth seal is that it must retain long term dimensional stability given the very tight clearance requirement between this dynamic seal and its mating part. A greater conclusion is reached regarding this mechanism failure when the design heritage of the damping cartridge is examined. It turns out that; a material other than Torlon was used in previous versions of the damping cartridge for the labyrinth seal. Therefore the true root cause for the SRTM damping mechanism failure is the damping cartridge design heritage was voided. An additional comment worth considering, is that a protoflight development approach was followed with this system based project schedule and cost constraints, and consequently specific engineering models were not developed. Hence in order to meet shelf life requirements, similarity to known existing designs. Unfortunately, the required similarity was lost due to a seemingly innocuous design change.

## 4. Conclusion

The mast vibration damping system implemented for SRTM was discussed herein. The system design followed a straightforward approach and used off-the-shelf components modified to meet specific performance requirements. Further, this system met all acceptance and performance test requirements, yet failed in-service, i. e. on-orbit. The technical root cause of the systematic failure of this system was identified and discussed. Additionally, a more general failure root cause was discussed. This data is presented in order to benefit future damping system applications. Specifically, the damping cartridge design discussed above, and used by SRTM, can be used with confidence provided the deficiencies associated with this failure are corrected.

## 5. Acknowledgments

The effort described in this paper was carried out at the Jet Propulsion Laboratory, California Institute of Technology, under contract with the National Aeronautics and Space Administration.

## 6. References

D. Gross, and D. Messner, 1999. "The Able Deployable Articulated Mast – Enabling Technology for the Shuttle Radar Topography Mission," Proc. of the 33rd Aerospace Mechanisms Symposium, NASA/CP-1999-209259, Pasadena, CA, pp. 15-30.

A. R. Klembczyk, and M. W. Mosher, 1998. “Applications of Hermetically Sealed Fluid Dampers for Low Level, Wide Bandwidth Vibration Isolation.” Proceedings of the 69th Shock and Vibration Symposium.

A. Stewart, C. Powers, and R. Lyons, 1998. “Improvements for Rotary Viscous Dampers used in Spacecraft Deployment Mechanisms.” Proc. of the 32rd Aerospace Mechanisms Symposium.